Diesel Railway Traction

The Progressive Spirit!

AS with all new inventions and devices, the diesel engine as applied to railway traction has had a hard uphill fight before any meed of recognition has been granted to it. Judging from the orders for 115 and 70 diesel railcars recently placed by the German and French railways, to say nothing of the 900-odd cars at work throughout the world, one might be forgiven for thinking that opinions on the latest form of traction were unanimous. But consider the report of the Queensland Government Railways for the year ending June 30, 1933. After emphasising that the greatest scope for railcar development in Queensland is in the improvement of branch line and country services, the Commissioner states his opinion that departure from the present petrol traction would not be justified until diesel traction development reached a more conclusive stage.

It would thus not appear to be known in the Antipodes that light branch line working by diesel-engined units has not only been carried on with consistent success for over twenty years, but has reached a higher pitch of perfection than any other phase of diesel traction, and in so far as operating expenditure, whether net or gross, is concerned, can show better figures than the majority of petrol cars. More than once we have mentioned in these pages that reliable information on the subject of operating results over a period of years is available for those who care to make use of it, and we have recorded the fact that something like a hundred light four-wheeled railcars with Ganz engines of 110-120 b.h.p. have covered over 10,000,000 miles on the Hungarian State Railways at a running cost including repairs and maintenance, but excluding wages of the crew, of 2·75d. a mile when hauling three light trailers. Petrol-engined cars engaged in hauling two trailers on similar services gave a corresponding figure of 3·9d. a mile, the cost of running maintenance being about 16 per cent. higher than with the petrol cars. This certainly does not confirm one of the reasons given by the Queensland Railways' Commissioner for not adopting diesel cars, that they are more costly to maintain. In addition to ordering only diesel cars for the past two years or so, the Hungarian State Railways are now converting their petrol cars to diesel vehicles as they come in for heavy repairs.

Nor is this an isolated case. The Swedish cars of 75 to 120 b.h.p., over periods up to 20 years, show total operating costs, including interest and depreciation, approximating to 11½d. a mile, or about 8¼d. a mile excluding these items. In Denmark, where the speed and mileage is somewhat greater, the operating cost a mile is even less, and annual mileages of 70,000 have been maintained by a number of vehicles since 1928-29. The ten 85-b.h.p. Renault cars of the French State Railways have covered between them over a million miles since being placed in service in 1931, and various light German cars, after having been at work since 1924-25 with 75-b.h.p. oil engines, are now being rebuilt with engines of twice the power. Unfortunately, the Queensland Government Railways are not alone in the way they view diesel traction, but although the diesel car manufacturer may suffer by lack of orders, a more serious aspect is that the railways are incurring an avoidable loss, and in the case of Government lines the loss is made good by the man in the street in the form of increased taxation.

Diesel Locomotive Flexibility

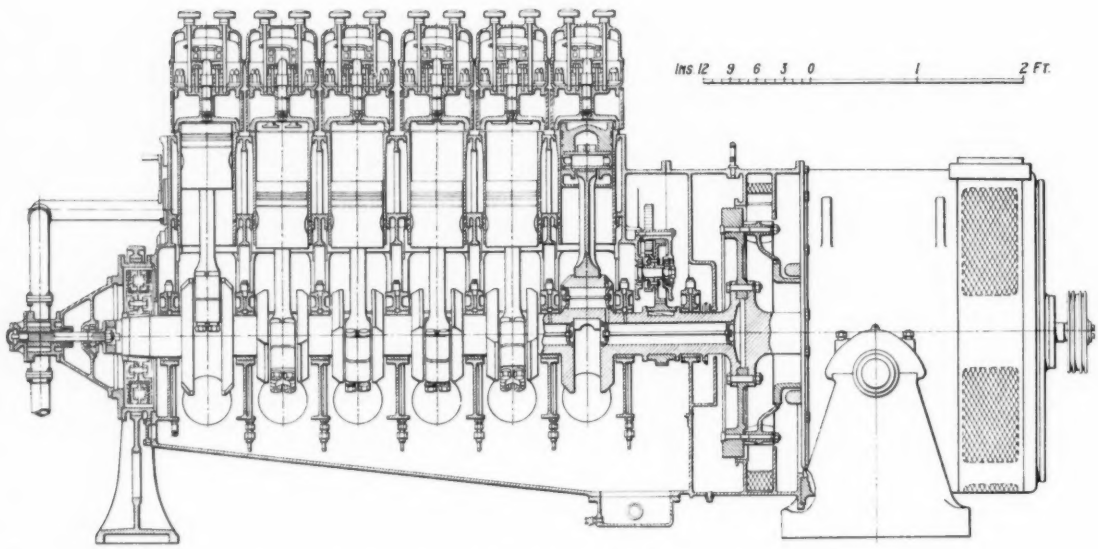
THE transmission question has, from the inception of the oil-engined railway vehicle, always been a thorn in the side of designers, and the cost, weight, and complication of the media for converting the engine torque to tractive effort at the rails has retarded the adoption of the diesel locomotive more than any other single factor. With the tradition of almost a century of steam locomotive practice behind them, it is not surprising that the designers of the first high-power diesel locomotive decided to adopt a form of direct drive; if successful, it would have met with the maximum of approval from railway officers, as a result of incorporating the minimum of unfamiliar apparatus. Unfortunately, the inherent disadvantages of the diesel engine in regard to starting under load and torque variation nullified the advantages of a simple transmission, and as the torque-speed characteristics of the diesel engine require a unit of at least 75 per cent. greater capacity than that required with an infinitely-variable transmission if equivalent railway-operating performance is to be provided, other forms of transmission hold the field.

Starting, and working at very low speeds, has always been a greater problem than flexibility in power at normal road speeds of, say, 15 m.p.h. and upwards. Although, theoretically, the amount of compressed air required to start the vehicle, and get the train moving at a speed at which the engine fires regularly, may be reasonably small, in practice it has proved to be large. In the Sulzer-Diesel locomotive of 1912 the auxiliary diesel engine developed 25 per cent. of the capacity of the main engine, and in the latest Deutz direct-drive passenger locomotive it is 20 per cent. Nevertheless, the attractions of direct-drive are great, and the difficulties of adequate torque at the lowest speed seem to have been partly solved in the Deutz locomotive by the adoption of a double-acting engine with a form of supercharging which does not incorporate the usual type of blower.

It is curious that the three purely direct-drive locomotives which have been produced in the last score of years should have been powered by two-stroke engines, which are not so flexible as their four-stroke counterparts. It is true that the power impulses are twice as many, but, among the problems of the direct drive locomotive, that of having to find sufficient room for a large number of cylinders has never been one. Some advance has been made by the Deutz engineers by adopting a double-acting engine—the first ever used in any phase of diesel railway traction. The layout of the Deutz locomotive is exceptionally simple, the cylinders being arranged very much as they are in a three-cylinder steam engine, but we suggest that the high-power diesel locomotive of the future will be fitted with a combination of pure direct drive for use at speed, and a form of electric or pneumatic transmission for starting and slow speed work.

AN UP-TO-DATE ENGINE FOR RAILWAY TRACTION

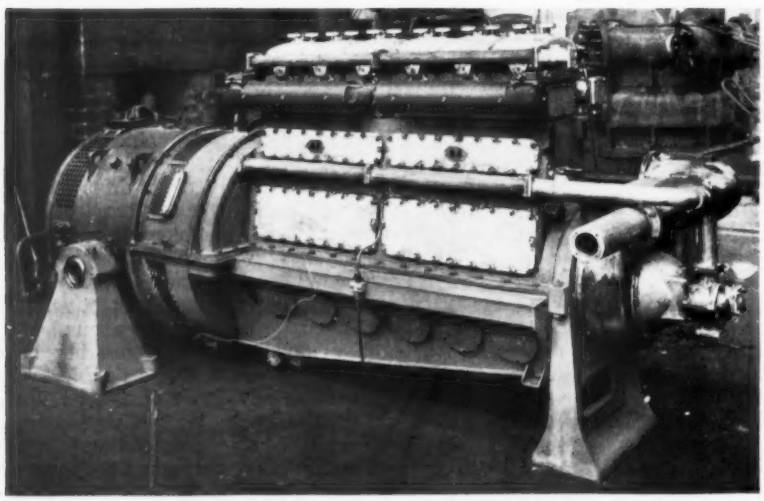
High-speed light-weight unit as used in conjunction with electric transmission for a British railcar



Longitudinal section of 200 b.h.p. English Electric diesel engine

IT is now being recognised that a light-weight high-speed engine is essential, rather than merely desirable, for railcar work, and one of the most recent designs is that fitted to the English Electric railcar described in the December 29 issue of this Supplement. Although the weight is such as to suit it for certain locomotive duties, the English Electric engine, which was designed at the Rugby works, is characterised by the possession of the highest piston speed yet attained in rail traction work, and by rotational speed exceeded in the same sphere only by the A.E.C. engines fitted to the Hardy and G.N.R. (I) diesel-mechanical vehicles.

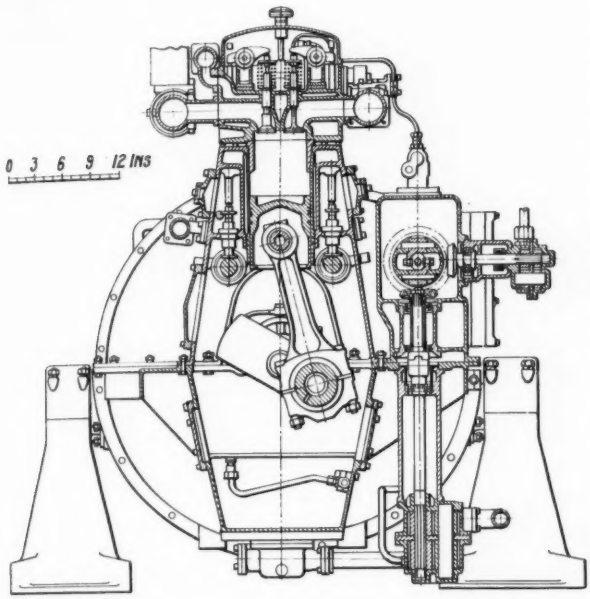
Longitudinal and cross sectional drawings of the engine are presented herewith, and it will be seen that six cylinders with direct airless injection are used. The cylinders have a bore and stroke of 6·0 in. by 8·0 in., and with the rated output of 200 b.h.p. at 1,500 r.p.m., the piston speed is 2,000 ft. per min. and the brake m.e.p. 78 lb. per sq. in. The engine fuel consumption and torque curves show that the brake thermal efficiency is high—36 per cent. under optimum conditions—and that the torque is well maintained over the higher range of speed. The engine alone weighs 4,900 lb., or with attached generator as shown in the accompanying illustrations, approximately four tons. The bedplate is of welded steel construction, and when coupled to the generator the unit is arranged for three-point suspension, with rubber blocks beneath the pedestals.



200 b.h.p. railway oil engine with generator attached

Cylinders and Crankcase

The cast steel crankcase and cylinder block are integral, and bored to receive the liners, which are of special cast iron with bores finished by honing. In order to permit of expansion and contraction, the liners are fixed at the upper end only, and the water joint at the lower end is formed by rubber rings having good oil and heat resisting properties. It will be seen from the general arrangment



Cross-sectional arrangement of 200 b.h.p. oil engine

drawings that the design of the crankcase and cylinder block not only gives a very rigid support for the main bearings, but ensures an ample water space between the cylinders. The cooling problem has been given further close attention in the cylinder heads, which are cast separately of a heat-resisting iron, to a design which, although unusual, is free from pockets, and enables an efficient circulation of the cooling water to be maintained. Circulation of the cooling water is effected by a centrifugal pump driven from the engine crankshaft. Both water and lubricating oil radiators are located in a single group on one side of the coach, and the electrically-driven radiator fan is arranged to run for approximately three minutes after the engine is shut down.

Four valves per cylinder are used, and these are driven from the cam shaft by push rods and rocker gear of normal design. The cylinder covers may be lifted off without disturbing the valves or rockers. Although valve spring failures are comparatively rare, two springs per valve have been incorporated in the design under consideration, and thus reduce the risk of failure to a minimum.

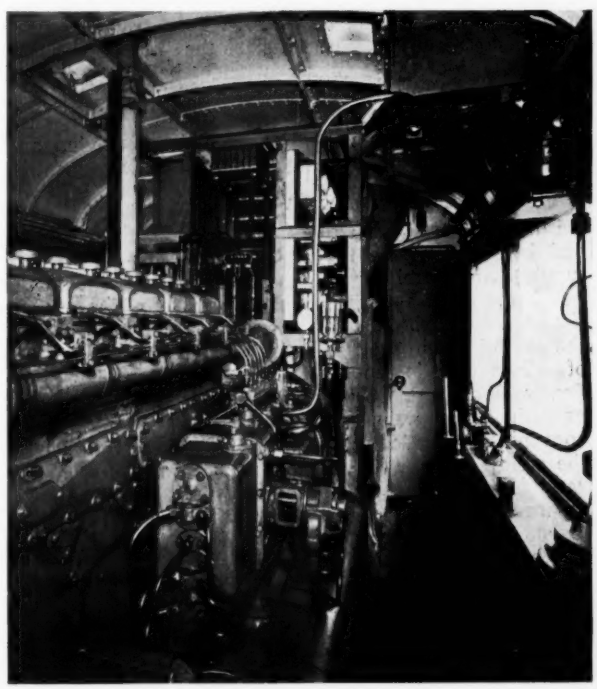
Carried in seven white-metal lined bearings, the six-throw crankshaft is forged in one piece and hollow-bored throughout, the ratio of the bore to the outside diameter being 0·52 for both pins and shaft. The dynamic loading on the bearings is reduced by prolonging the crank webs to form a partial balance. A flange is forged on one end of the shaft, and to this the flywheel is bolted. The connecting rods are drop stampings of 35-ton carbon steel, with white-metalled big-end bearings and phosphor-bronze bushes for the gudgeon pin. Aluminium alloy pistons, carrying one scraper and three pressure rings, are used, and the crowns are shaped to provide the maximum practicable distance between the central injection nozzle and the nearest surface which can be reached by the fuel spray.

Governing and Fuel System

The centrifugal governor is mounted on an extension of the fuel pump driving shaft, and is designed to operate at a number of fixed speeds, which, in the case of the railcar now running on the L.M.S.R., are 900 and 1,200 r.p.m. in addition to the full load speed of 1,500 r.p.m. Energised by the master controller of the railcar, a number of solenoid-operated valves are arranged to admit lubricating oil, under pressure from the main system, to control cylinders in which are run spring-loaded pistons connected through the springs to a layshaft. When oil is admitted, the piston moves against a stop, thus compressing the spring and externally loading the governor by the desired amount. The layshaft is also connected to the fuel pump control rack, and regulation of the fuel is carried out by the movement of the rack rotating the fuel pump plungers, and thereby determining the quantity of fuel delivered per stroke. Any failure in the lubricating oil system, which operates at a maximum pressure of 36 lb. per sq. in., results in the main engine being automatically shut down. The timing of the injection is automatically advanced in conjunction with an increase in the rotational speed of the engine, and efficient combusion is thus ensured over the whole speed range. The fuel pump itself is of the six-ram type, with Bosch elements, the pump block and attachments being made by the English Electric Company.

Lubrication

A gear-type pump, driven from the crankshaft, lifts the oil from the sump tank and delivers it through a fine-mesh strainer and an oil cooler. The oil passes to a bus-pipe inside the crankcase, and connections from this pipe feed the crankshaft main bearings. The crankshaft is drilled to supply oil to the connecting rod big-end bearings, from which it is led up through the rods to the small-end bearings and floating gudgeon pins. The pistons are mist lubricated by oil vapour from the crank chamber, and the valve gear, camshaft bearings, and all other points are supplied with oil from the pressure lubrication system. To maintain the oil in clean condition, a small portion of it is bye-passed from the pressure system after it has passed through the strainer, and this portion is led through an oil cleaner of felt, back again to the sump tank. Both cooling water and lubricating oil systems are protected by Monitor devices, and the latter system is limited to a maximum pressure of 36 lb. per sq. in.



Engine room from driving compartment

CONTROL FOR DIESEL-ELECTRIC VEHICLES

Constant-power characteristics of simple hydraulically operated control system evolved by British firm

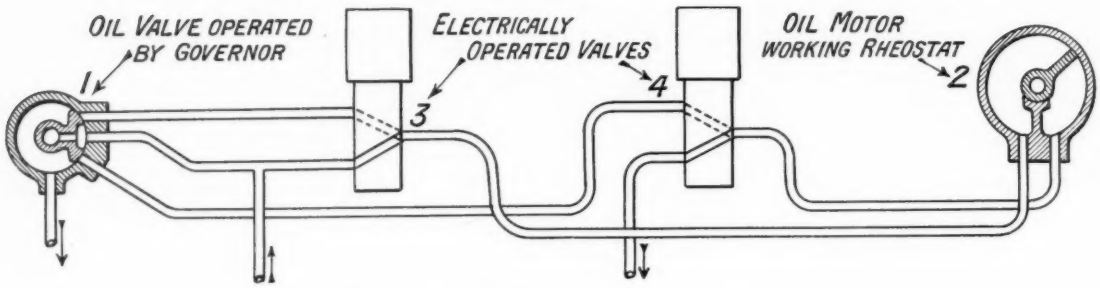


Diagram of hydraulically-operated control gear

MOST early forms of automatic control for diesel-electric vehicles took the form of a de-compounded generator, that is, a generator with windings such that with heavy currents the voltage fell away to zero. It is possible to design the windings so that the product of volts and amperes at no point over the whole current range of the machine shall exceed a desired value, corresponding to the rated horse-power of the engine. Unfortunately, it is at one point only on this current curve that the rated horse-power of the engine is at all fully employed, and though this can be improved by complications in the motor control equipment, the result, considered over a normal start to stop journey, is a poor engine utilisation.

In view of this defect in the de-compounded generator form of control, Sir W. G. Armstrong, Whitworth & Co. Ltd. set themselves, three years ago, to the perfection of a scheme of automatic constant horse-power control. Sulzer Bros. Ltd., to whose designs the Armstrong-Sulzer railway type of diesel engine is made, had employed an excellent system of manual control which gave the results desired, but it was realised that for use in different parts of the world and with various methods of driving and supervision, a very simple automatic control was essential.

Systems Investigated

Different media—electricity, compressed air, vacuum, and hydraulic (using the engine lubricating-oil pressure supply)—was carefully investigated, and tried out on the test bed. The hydraulic system was found to be the simplest, most reliable and most easily maintained; it gives the control within the fine limits desired, and the bugbear of all automatic control systems—hunting—has been finally eliminated. Oil was proved to have advantages over the other systems of having its source of pressure conveniently situated, *i.e.*, on the engine itself; of allowing the control to give instantaneous response to the varying conditions of train speeds, grades, &c.; and of easy maintenance owing to the lubricating qualities of the oil.

One of the greatest advantages of the Armstrong-Whitworth system is that it does not depend upon definite movements which would entail very careful maintenance. It is self-adjusting against wear and leakage of oil and against varying conditions of the temperature of the oil, and of the main generator and other apparatus. The apparatus was fitted to the diesel-electric railcar *Tyneside Venturer*, then undergoing works' running tests, and was considered suitable in every way for the duties it had to perform. It was later used on shunting locomotives in 24-hr. service, and is now standard for all Armstrong-Sulzer rail vehicles.

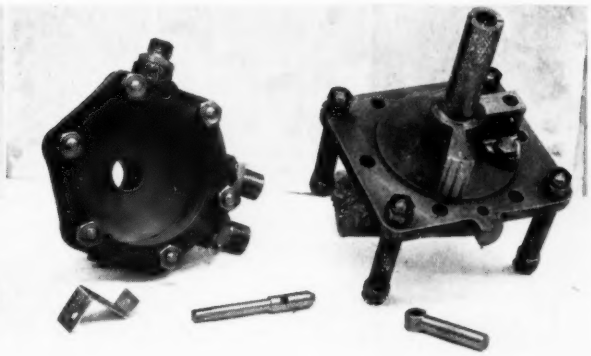
Features of System Adopted

The main features are shown in the accompanying diagrammatic sketch, and the two items of the equipment are illustrated by photographs. By means of an oil valve (1) oil under pressure is admitted to one or other side of an oil motor (2). The oil valve is mechanically connected to the engine governor and responds to any slight change in the position of that instrument; the oil motor is mounted at the back of a rheostat, to whose spindle it is directly coupled. This rheostat controls the voltage of the main generator, from zero when the vehicle is starting, and from rest up to the maximum voltage when the vehicle is travelling at its full speed.

There is, of course, a certain position of the governor corresponding to full rated fuel admission; any slight overload causes a fall of this governor position, and a slight underload causes a rise. These small movements are transmitted to the oil valve, which then admits oil from the central pipe to the one immediately above or below. The pipe leading vertically down from the bottom of the valve is merely an oil drain back to the sump. The rotor of the oil motor responds to pressure in either of these pipes, turns in the required direction, and takes load off the engine or puts additional load on, by lowering or raising the voltage of the generator. This is immediately answered by the return of engine speed to normal, the ports in the oil valve closing, and the oil motor remaining in this position until a further change in train speed.

The Armstrong-Sulzer engine is designed with two or more normal running speeds, corresponding to full load and fractional loads, and change in the engine speed is effected by control of the governor spring pressure. The position of the governor itself, therefore, remains the same at whatever speed the driver selects, and the automatic control consequently functions in exactly the same manner as described and maintains the engine at full, or whatever fractional power has been chosen. The two electrically-operated valves, 3 and 4, provide what may be termed supervisory control. As shown in the diagram, these valves are de-energised, the dotted lines showing the oil circuits in normal operation when both valves are

Oil valve operated by governor



Oil motor working rheostat

energised. The valve 4 prevents the rheostat from moving forward too fast at the moment of starting the train. It will be realised that in addition to limiting the voltage on account of engine output, it is also necessary to limit the voltage at the moment of starting to prevent an excess of current slipping the wheels. This is effected by means of a current-limit relay acting through valve 4. Valve 3 is normally energised when the train is in motion, but when the driver cuts off power this valve is de-energised and causes the rheostat to move right back to the zero voltage position ready for the next train start. No lubrication of parts is necessary, since each element is working in oil, and maintenance is therefore negligible; slight leakage of oil has no serious effect because the movements are self-correcting, and in practice it is found to control the engine horsepower within 2·5 per cent. over the fuel working range of the vehicle. A further advantage of this type of power control is that in the event of one cylinder out of six having been cut out, say on account of a faulty fuel pump, the apparatus will automatically keep the engine output at five-sixths of normal output.

The device has, together with additional apparatus for effecting requirements of control under certain conditions, been protected by patents in all parts of the world. Among the modifications which may be embodied may be mentioned the incorporation of the two delivery ports of the oil valve in a sleeve situated between the oil valve proper and its stator. Rotation of this sleeve through a few degrees allows the mechanism to be set for any desired percentage above or below normal full load. All Armstrong-Sulzer railway vehicles for passenger service are fitted with remote control of this sleeve, allowing the overload capacity of the engine to be utilised when required, at the will of the driver.

RECOGNITION OF THE DIESEL

An Argentine railway manager on the future of the internal-combustion engine

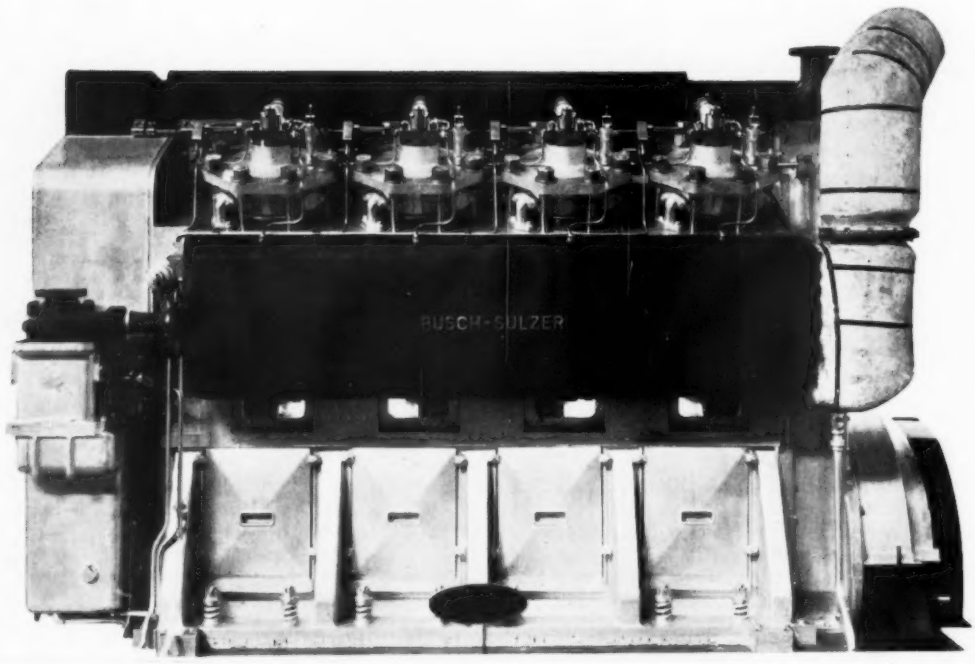
AT the annual dinner of the South American Centre of the Institution of Locomotive Engineers held in Buenos Aires a short time ago, Mr. M. F. Ryan, General Manager of the Buenos Ayres & Pacific Railway, delivered an interesting address on the future of the internal-combustion engine and its probable influence on railway traffic, both steam and electric, in the course of which he mentioned that of the 22,000 miles covered during his recent travels in Europe, 18,000 had been by means of the internal-combustion engine, and ten per cent. of the 4,000 miles traversed by rail had been in internal-combustion railcars. On this record, supplemented by his own observations, Mr. Ryan based his conviction that the internal-combustion engine is rapidly shouldering out both steam and electricity in transport, and he reminded his audience that although from 1840 to the end of the 19th century, steam had virtually held a monopoly as the motive power for mechanical transport by road, rail and sea, whole provinces were now being wrested from the domain of steam; electric traction was nearing the end of its heyday, and the internal-combustion engine was making triumphant progress in all branches of transport.

Mr. Ryan considered that the express steam locomotive had another enemy in the fast railcar, apart from aircraft, which would probably kill the long-distance European luxury express. This vehicle was still only in the elementary stage, however, and it was not yet certain whether the Flying Hamburger and Micheline types of car, which required a deal of nursing, would develop to maturity or else go the way of the locomotive turbine. One thing, however, was certain; for branch-line work the petrol or oil-driven railcar would oust the local steam train. The Hungarian State Railways (on which he had travelled) had 100 of these cars in service, and 100 more on order or projected. All the new cars had diesel engines, the operating and maintenance costs being half what they were with steam. In his opinion, the internal-combustion engine, and more especially the oil engine, up to 300 b.h.p., would replace steam for light work on all railways.

In the sphere of electric traction Mr. Ryan pointed out that the high-powered, double-decked bus was superseding the electric tramway everywhere in Britain. Motoring through the country, it was surprising the number of towns where there were tramlines but no trams, and he ventured to say that there must be very few places in the world where tramways were being built or even extended. Some electrical undertakings were compromising by operating trolley-buses; but this was a half-hearted measure, as such vehicles could never compete with the flexibility of the internal combustion-engined bus. Mr. Ryan said he would not be surprised if we never saw another big railway electrification scheme go through. For dense suburban work, especially in tunnels and on mountain railways where water power is available, electricity might hold its own, but the big diesel had sent out the challenge and, to his mind, might win the day.

THE LARGEST LOCOMOTIVE DIESEL ENGINE

New American design develops 1,600 b.h.p. in eight cylinders



1,600 *b.h.p. two-stroke Busch-Sulzer diesel engine*

WE have more than once commented upon the growing necessity for a diesel engine suitable for railway traction which would develop an output of 1,500 b.h.p. or more, and suggested that this could be met only by the adoption of the two-stroke principle. It is therefore gratifying to be able to record the construction of an engine producing 1,600 b.h.p. at the normal rating. As might be expected, this design was developed in the U.S.A., where the locomotive powers required for anything but shunting work are far beyond the capacity of the existing railway-type oil engines.

The first engine to this design has recently been undergoing bench tests at the works of the designers and builders, the Busch-Sulzer Bros. Diesel Engine Company, of St. Louis. It is of the V type, with eight two-stroke single-acting cylinders 13·5 in. dia. by 16 in. stroke, and gives an output of 1,600 b.h.p. at 550 r.p.m., the corresponding piston speed and brake m.e.p. being respectively 1,465 ft. per min. and 63 lb. per sq. in. The weight of the engine is slightly under 25 lb. per b.h.p., and the design provides for the addition of further pairs of cylinders up to a total of 16 cylinders developing 3,500 b.h.p. at 600 r.p.m.

Cylinders and Crankcase

The engine main frame and cylinder block are integral, and are cast of iron or aluminium alloy, depending upon the weight limitations. The frame has large but light oil-tight inspection doors on each side. An oil trough is attached to and below the frame, and the working parts are thus fully enclosed in an oil and vapour-tight structure. The block and liners are so arranged that each cylinder has a sludge chamber near its lower end. This chamber is fitted with upper and lower oil wiper rings through which the piston works and which prevent dirty oil, sludge and sparks, from the combustion end of the cylinder, from passing into the interior of the engine frame or crankcase, and oil from the crankcase from being carried into the upper part of the cylinder. Each sludge chamber has an opening on the outer sides of the engine protected by a glass cover through which the piston may be observed while the engine is running, and its proper lubrication thus assured—avoiding excessive as well as insufficient lubrication. The sludge chambers are electrically lighted.

That portion of the block situated between the two rows of cylinders serves as a distributing receiver for the scavenging air, and in it are located the automatic scavenging valves, attached to the faces of the block. The top of this space is closed in by the housing of the scavenging blower, which extends almost the full length of the engine. The air inlet duct to the blower runs along the top of the housing. In a chamber, at the end of the frame farthest from the generator, are contained the fuel injection pumps, the governor and its mechanism, and the cylinder lubricating pumps.

Moving Parts

The crankshaft is a single steel forging, of heat-treated steel. Each crankpin serves for two opposite cylinders, so that the number of cranks is half that of cylinders. An

integral coupling flange is forged on the generator end of the crankshaft. The main bearings consist of upper babbitt-lined half-shells, in seats formed in the frame, and lower similar half-shells carried in substantial forged inverted caps, which are fitted into the frame and supported by heavy bolts of alloy steel, the shells themselves being forged of aluminium.

The connecting-rods are heat-treated open hearth steel forgings, and are bored throughout their length, for the passage of lubricating oil to the gudgeon pins. The gudgeon pins are of hardened steel, ground and polished on their working surfaces. Each pin is rigidly bolted to the flange at the top end of its connecting rod, the flange resting against a flat surface milled into the power side of the pin.

The assembled gudgeon pin housing and piston form a single structure of aluminium alloy. The housing is attached to the inside of the piston, and incorporates the pin bearings, so that the piston itself is free from the heavy bosses usually required for the attachment of the pin, and the walls of the piston skirt are not pierced by any holes. Thus all transverse sections of the piston proper are concentric, symmetrical, closed circles, avoiding distortion and leakage of lubricating oil to the cylinder. All pistons, housings and connecting rods with their bearings and pins throughout the engine are alike, the axes of the companion cylinder pairs being offset to permit this.

Exhaust and Scavenging

Along the front of each line of cylinders extends a water-jacketed exhaust header, into which the cylinders on that side exhaust. Oil-catching and spark arresting provisions are connected to each header, and are arranged to drain to the roadbed. The exhausting, scavenging, and fresh air charging of the cylinders are performed through ports in the cylinder well. The exhaust ports are controlled by the pistons, and are located around one half of the circumference of the cylinder, communicating directly from the interior of the cylinder into the exhaust header.

Scavenging and charging are on the Sulzer system, with two tiers of ports round one half of the cylinder circumference, opposite the exhaust ports. The upper tier, which is uncovered by the piston before the exhaust ports are uncovered, is controlled by automatic valves, opening towards the interior of the cylinder. These valves are located in the crotch receiver, as previously described, and prevent a blow-back of exhaust gases into the receiver, as they remain closed until the pressure in the cylinder has fallen below that of the air in the receiver. The scavenging ports in the lower tier form a direct communication from the receiver into the cylinder, and are uncovered by the piston after the exhaust ports have been so uncovered, and when the pressure in the cylinder has fallen well below that of the scavenging air. On the return stroke of the piston the scavenging ports in the upper tier remain open after the exhaust ports have been covered by the piston, and the flow of air, for the charge, continues until these scavenging ports have been so covered, the cylinder thus being filled with fresh air at a pressure above that of the back pressure of the exhaust. This effects a moderate degree of supercharging, without separate machinery for this purpose. Scavenging air is furnished by a blower of the rotary positive-displacement type, with top suction and bottom discharge. It is driven through gears and a flexible coupling from the crankshaft. The air passes through filters and a muffler in its passage to the blower, and is discharged directly into the crotch receiver.

Fuel System

The fuel injection system is of the A.E.G.-Hesselman type, and comprises a fuel booster pump, a fuel injection pump, fuel filters, and fuel valves. The pump is under governor control, and measures and delivers to the fuel valves the quantity of fuel appropriate for the existing engine load and speed. The pump plungers are operated by hardened steel cams on a shaft that is driven from the engine crankshaft, through gears. The quantity of fuel delivered to the cylinder at each delivery stroke of the pump is regulated by the point of opening of a spill valve on the pump. This point is varied by the engine governor, being earlier when the governor collar is in a high position, *i.e.*, when the load is light and little fuel is needed, and later when the governor collar is in a lower position, and the load heavier. Thus the first portion of the plunger discharge is that which goes to the working cylinder, and the fuel that is released by the spill valve returns to the fuel supply line of the pump. To ensure the filling of the pump barrels at each suction stroke, the fuel supply is put under moderate pressure by a booster pump, and the delivery pressure of the fuel injection pump is about 4,500 lb. per sq. in.

Fuel is delivered, by each plunger of the fuel pump, to its individual fuel valve, passing on its way through a fine filter. The fuel valves are of the A.E.G.-Hesselman membrane-spring type, and communicate with the spray nozzles in the cylinders. The valves are opened by the pressure of the fuel oil as soon as the respective pump plunger begins to deliver fuel, and are closed by the membrane springs as soon as the oil pressure is relieved by the opening of the spill valve. The shape and rotative position of the fuel pump cams therefore control the timing of the commencement of injection into the cylinders.

In engines operating at constant speed, the fuel lead, *i.e.*, the point ahead of top dead centre at which the injection of fuel begins, is constant; but this engine, intended to operate at any speed between half and full, is provided with an automatic device for varying the fuel load to suit the engine speed, increasing it at the higher speeds and diminishing it at the lower, and thus preventing the violent ignitions and dangerous rises in pressure that would occur if the time period for ignition before top dead centre were unduly lengthened. This makes it possible to maintain substantially constant full torque at all engine speeds. The fuel is sprayed through the atomising orifices of the spray nozzle into the combustion chamber, which is of the A.E.G.-Hesselman contour, where it is ignited.

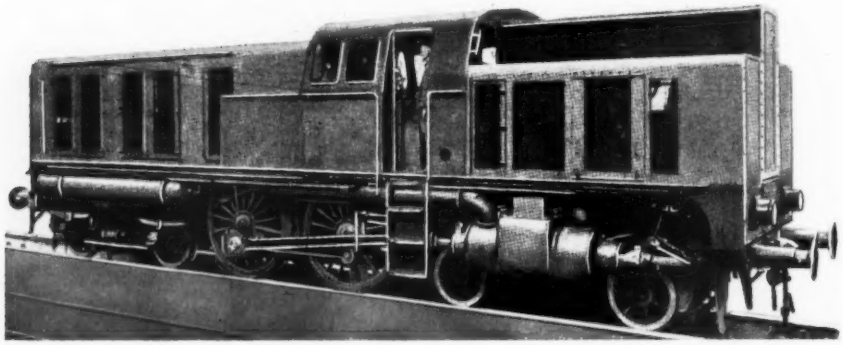
A centrifugal governor controls the engine, and maintains the speed at that for which the governor is at the time adjusted, *i.e.*, anywhere between half and full speed. The adjusting of the governor is accomplished by a hand operated device, which permits the variation of the operative speed of the governor from a remote point. Thus the engine may be run at the speed that is most advantageous for the existing locomotive speed and load. The driver's control of the engine speed is incorporated with the control of the traction motors where electric transmission is employed.

Lubrication

The general lubrication of moving parts of the engine is on the pressure system. The oil flowing from the lubricated parts, and including the clean oil wiped off by the lower rings in the sludge chamber, is gathered in the oil trough under the engine, whence it is re-circulated after being filtered and cooled. There are two centrifugal pumps for circulating the lubricating oil, one to draw the oil from the trough and deliver it to the coolers, the other to draw it from the coolers and deliver it, under pressure, to the engine bearings and other parts requiring lubrication. Separate mechanical lubricators are provided for the lubrication of the cylinders, and these are automatically kept filled with clean oil.

DIRECTLY-DRIVEN DIESEL LOCOMOTIVE

High-power supercharged German machine now undergoing running trials shows encouraging results



Direct-drive diesel locomotive for light passenger service

DESPITE the attractiveness of a direct drive between the engine and the road wheels, the inherent defects of the diesel engine are such that only three attempts have been made to utilise this form of transmission in the 22 years' history of the oil-engined locomotive. The first two machines, the Sulzer-Diesel 2-B-2 locomotive of 1912, and the Ansaldo 2-C-1 unit of 1929, embodied two-cycle engines and thereby sacrificed a certain amount of flexibility in order to gain simplicity, and the third example of direct drive, found in the locomotive now being tested by the Humboldt-Deutzmotoren A.-G., which firm constructed the unit to its own designs after experimenting with a smaller machine, is also powered by a two-cycle engine. Another form of direct drive is the Kitson-Still, but in this case the diesel cylinders are assisted by steam on the other side of the pistons at starting and during periods when great power is required.

The 2-B-2 wheel arrangement, the two-stroke engine, and the rated output of 1,000 b.h.p. of the Deutz product are the same as those of the pre-war Sulzer machine, but there the similarity of the two designs ends, for the Deutz locomotive has three double-acting cylinders arranged in the same fashion as those of a steam locomotive, in place of a central crankshaft driving four single-acting cylinders arranged in a V. The 82-ton Ansaldo locomotive was powered by a six-cylinder two-stroke opposed piston engine mounted in the cab structure, the scavenging pump cylinders, which were used for starting, taking the place of normal locomotive cylinders.

Direct Drive Difficulties

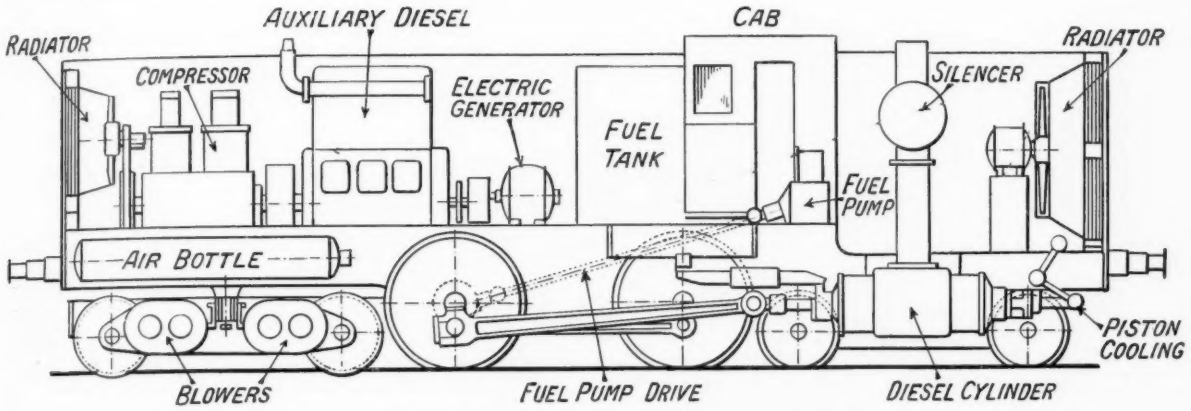
One of the basic problems of the diesel locomotive is that of developing adequate torque at low speeds. Rigid connection of the motor with the driving axles is practicable only with a working cycle which develops, by aid of the smallest possible amount of external energy, sufficient torque at low speeds to provide for starting under load, acceleration and running up hill. After several years' research work the Humboldt-Deutzmotoren A.-G. developed the diesel engine so that it became possible to burn oil in the starting air during the first revolutions of the driving axles, without pressure-rise. Acceleration and climbing power are secured by supercharging; and the ordinary diesel cycle with airless fuel injection suffices for running on the level. The compressed air requirements of the locomotive are thus reduced to such an extent that the dimensions of the auxiliary engine and compressor remain within reasonable limits, viz., about 20 per cent. of the normal indicated power of the locomotive cylinders.

Early Trials

Preliminary trials of supercharging for climbing showed that it was impossible to start under load with cold, high-pressure air and change over to supercharged diesel operation after reasonable consumption of compressed air. The repeated expansion of cold air so chilled the walls of the combustion space that high acceleration was required to ignite the injected fuel. Pre-warmed high-pressure air gave little better results, the air quickly losing most of its heat, and the subsequent expansion again resulting in such cooling that ignition could not be relied upon at low speed. Trials of various methods of warming the compressed air led to the development of a device for burning small quantities of oil in the compressed air at constant pressure and without risk of explosion. In the first instance, an electrically-heated glow-coil was used to maintain this combustion in a separate chamber, but it was soon found practicable to effect it in the engine itself.

Following upon successful trials with a stationary engine, the investigations were continued with a standard Deutz four-cylinder geared diesel locomotive of 100 b.h.p., retaining the clutch and using the top gear. Satisfactory starting under load was obtained and the consumption of air was quite moderate. On the other hand, the backlash of the gears accentuated the shocks resulting from irregular ignition during the starting period, and there were also shocks when coasting. These difficulties were overcome by eliminating gears and driving directly on to the axles by means of connecting rods, as in steam locomotives; but with this arrangement it was necessary to avoid peaks of combustion pressure, and even to decrease the normal pressure in order to reduce the stresses on the motion work.

In order to try out starting with reduced pressures and increased admission, the injection and valve gear of the 100 b.h.p. locomotive were reconstructed. In addition to the injection nozzles and pumps for ordinary diesel



Layout of new 1,000 b.h.p. Deutz direct-drive locomotive

working, the cylinders were fitted with a second injection device for starting at low pressure. The range of admission for air and fuel at starting was increased to 100 deg. of crank angle; and the ratio of injected fuel to air supply was controlled to ensure regular firing. Exhaustive tests on a horizontal, double-acting, two-stroke cylinder provided the basis of design for the actual locomotive cylinders, and the construction of the experimental locomotive, which embodies buffing and drawgear and various other standard details of the Reichsbahn, is shown by two of the accompanying illustrations.

1,000 b.h.p. Locomotive

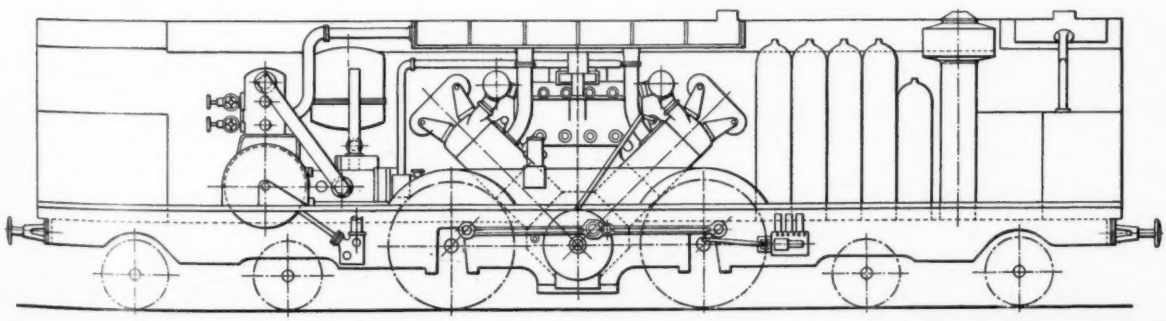
This locomotive was designed for light express duties, where about 600 d.b.h.p. is required in normal work, and the maximum speed does not exceed 68 m.p.h. The three double-acting cylinders are each 300 mm. (11⅞ in.) dia. by 600 mm. (23⅝ in.) stroke, the outside cylinders driving on to the crank pins of the second pair of coupled wheels, and the inclined cylinder on to the crank axle of the leading pair of drivers. The driving wheels are 1,750 mm. (5 ft. 8⅞ in.) dia., and the locomotive weighs 78·75 tons in working order, of which total 33·5 tons are available for adhesion. At the maximum road speed the driving wheels make 333 r.p.m. This proportion is low, but it was desired to keep the axle load to a maximum of 17 metric tons. By allowing the bogie pivots a side play of 50 mm. per side, curves of 395 ft. radius can be traversed. The framing is a combination of the bar and plate patterns, and by suitable designing, the stresses in the rods, crossheads, axles and cranks have been kept within the limits of modern steam practice without making the details cumbersome.

As may be seen in the diagram, the supercharging blowers for the driving cylinders are gear-driven from the axles of one bogie, the air being led through ducts to the air bottles and cylinder block. The main engine pistons are cooled by water (an unusual medium), the pump being driven through an arrangement of links connected to the piston tail rod.

Auxiliaries

The auxiliaries are driven by a vertical three-cylinder single-acting four-stroke engine, with cylinders of 240 mm. (9⅜ in.) dia. by 420 mm. (16⅜ in.) stroke, which is mounted in the cab, and drives the compressor, one radiator fan, and an electric generator. Radiators for the engine circulating water and lubricating oil are arranged at each end of the locomotive, that at the end farthest from the auxiliary diesel engine being driven by a small electric motor without the intermediary of gearing or a belt. The cab is situated near the centre of the locomotive and has controls at each side. The fuel pump and its control gear are also situated in the cab, the drive being taken from the second driving axle. Large bonnets at each end of the cab house the auxiliary apparatus, fuel tank and silencer.

The locomotive left the Deutz works over seven months ago and has since been running experimental road trials on different stretches of the German State Railway. Although various troubles have been encountered, it is understood that the results have, generally speaking, been encouraging, particularly as regards the special features connected with the starting and direct drive. From the vehicular point of view the locomotive seems to be all that could be desired, although the 4-4-4 wheel arrangement has in some instances been subject to criticism on the score of lack of stability. The trials are of considerable importance, for a successful direct-drive locomotive would probably create a much larger demand for diesel locomotives than at present exists.

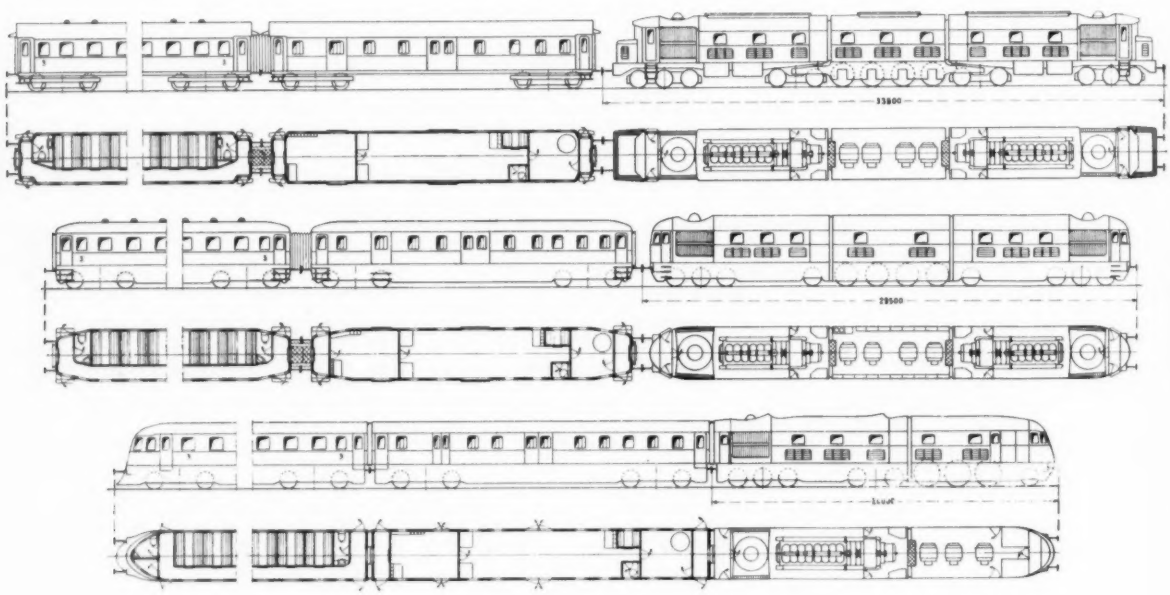


Layout of 1,000 b.h.p. Sulzer direct-drive locomotive, 1912

POSSIBILITIES OF HIGH SPEED

Streamlined trains hauled by powerful diesel-electric locomotives should show reduced operating expenditure and improved service

THERE is little doubt that a vast market awaits the high-speed diesel train, whether in the form of a locomotive and coaches or as a triple-articulated train set, for it can provide a cleaner, faster, and more economical service than can be attained with steam traction. In the range of really high speed, air resistance is the dominating factor in the exterior design of the train, and with trains of normal construction it increases by well over 100 per cent. with an increment in speed from 60 to 95 m.p.h. Much attention has recently been given in Britain, America and on the Continent to the possibilities of streamlining, and some of the advantages to be gained in operation by this form of construction are given detailed consideration in this article, and are based on the design of streamlined and semi-streamlined locomotives and trains evolved by Sulzer Brothers Ltd. as a result of extensive study.



Type 1—Normal train; diesel-electric locomotive of 3,900 b.h.p.
,, 2—Improved shape; ,, ,, ,, ,, 3,000 ,,
,, 3—Fully streamlined shape; ,, ,, ,, ,, 2,300 ,,

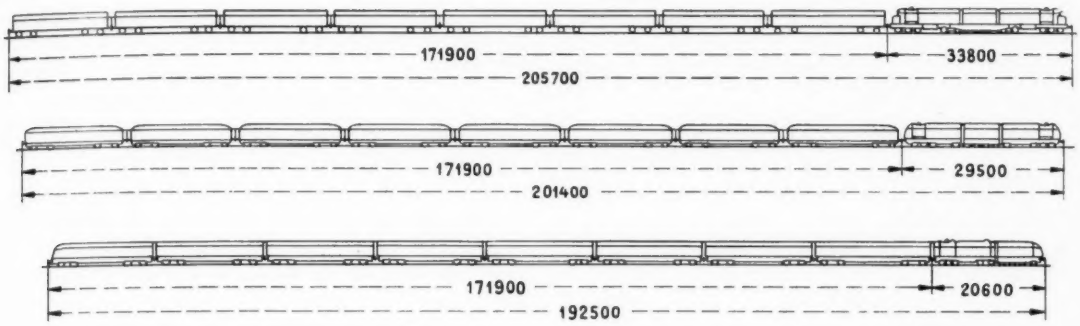
Fig. 1—High-speed trains with diesel-electric locomotives for a maximum speed of 80 m.p.h.

Figs. 1 and 2 show three trains with practically equal accommodation and designed for the same speed, namely, 80 m.p.h., the first train being of orthodox construction, the second somewhat improved as regards shape of vehicles, and the third a train streamlined as far as is practicable. With regard to the second train, the locomotive and coaches, including the rear one, can run in either direction, as the shape of the ends of the locomotive and coaches have been only partially streamlined. End and side panels are extended downwards below the frame in order to shield the bogies and intervening space, the design embodying, from considerations of air resistance, improvements over the usual form of construction shown in the first example.

The third train is one of fixed composition and is intended to run in one direction only. At terminals, therefore, it will be necessary to turn the whole train round, or alternatively shunt the locomotive and end coach, an operation which it is claimed offers no particular difficulty. With this train arrangement it is possible to give a good streamline contour to the front of the locomotive and to the end of the rear coach. At the same time care has been taken to keep the distance between the locomotive and the leading coach, and the spaces between intermediate coaches, as short as possible, although retaining standard buffing and draw gear. Side walls are extended downwards as in the case of the second train, but as a further improvement they are carried beyond the buffer beams and the vestibules brought out in line with the side panels, thus greatly reducing air resistance. Every precaution has been taken to avoid unnecessary projecting parts, the sides and roof being made as smooth as possible, usual type roof ventilators for example being replaced by the more up-to-date method of forced ventilation in conjunction with air-conditioning plant.

Air Resistance Tests

Recent work on the measurement of air resistance with streamlined trains has been published in the Transactions of the American Society of Mechanical Engineers for September, 1932. The tests were carried out with models



Type 1—Train weight, including locomotive, 600 tons
" 2— " " " " 560 "
" 3— " " " " 510 "

Fig. 2—Express trains with diesel-electric locomotives for a maximum speed of 80 m.p.h.

in a wind tunnel, and although they do not represent actual conditions in service, the results at least provide a means of comparing constructional forms. The following formula is adopted for air resistance:—

$$\text{Air resistance} = c\,\frac{\gamma}{2g}\,A_1\,V^2, \text{ where}$$

V = train speed in ft. per sec.
A_1 = projection of the cross-section of the train on a plane normal to the direction of running, in sq. ft.
c = coefficient of resistance of the train.
γ = specific gravity of the air in lb. per cu. ft.
g = acceleration due to gravity, 32·2 ft. per sec. per sec.

The coefficients of air resistance were determined by experiments with models of trains of both ordinary and streamline construction, values being recorded for the locomotive alone, and for trains of varying composition. The results of these experiments must, however, be accepted with reserve for the following reasons. The eddying between the coaches and the ground could not be determined in the wind tunnel, and the influence of side wind was not investigated. The surface roughness of the walls has a great influence on measured results, although in actual service this factor will be of less importance. Moreover, on account of the scale effect it is questionable whether the coefficients obtained with small scale models can be employed with confidence in calculating the air resistance of actual trains. However, the results of the investigation cannot be directly applied to the trains shown in Fig. 1, since in the American tests the ends of both the locomotive and the rear coach were flat, whereas in the proposed designs it will be appreciated that good streamline forms are considered of great importance.

Fig. 3 is reproduced from the A.S.M.E. Transactions, the left-hand diagram showing the coefficients of air resistance of a locomotive with trains of one to six coaches of the usual type, the right-hand diagram giving the coefficients for streamlined locomotives and trains. From this illustration the coefficients of air resistance for the three trains shown in Figs. 1 and 2 have been calculated and the results incorporated in Fig. 4. It will be noted that there are certain differences between the train of the first type shown, Figs. 1 and 2, and trains of conventional design, but the same resistance has been assumed. Figs. 3 and 4 present in striking fashion the rapid increase (proportional to the square of the speed) in the air resistance with a rise in the locomotive speed. The second type train does not possess such a good shape as the American streamlined train—in the latter the connections between the coaches were smooth—and it has therefore been assumed that the resistance of a train constructed on the lines of type 2 would have a value somewhere between the air resistances of ordinary stock and the American streamlined train. As far as the third type of train is concerned, the resistance in this case would be less than for the American train owing to the improved streamline shape of the rear coach, and for this reason the air resistance of the locomotive is assumed the same as in the American tests, whilst for the train with one to eight coaches it is reduced to some extent.

Three Types of Train

In order to show how the savings in power per unit weight—rendered possible by introduction of streamline construction—increase with increasing speed, the weight and power required for the three trains shown in Figs. 1

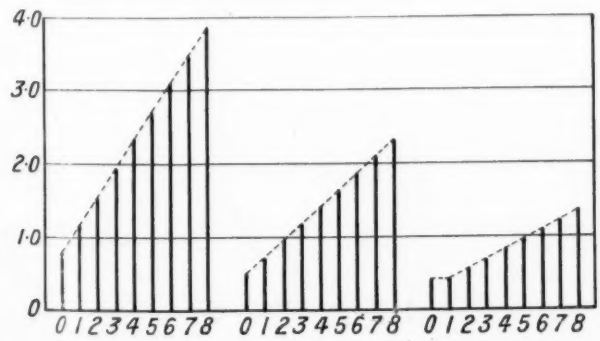


Fig. 3—Coefficients of air-resistance, A.S.M.E. formulæ

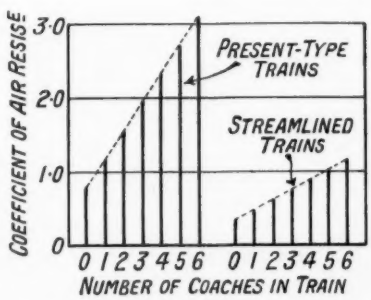


Fig. 4—Coefficients of air resistance for trains in Figs. 1 and 2

and 2 are given in Table I for speeds of 68, 80 and 93 m.p.h.

TABLE I

Train type		1	2	3
Maximum speed, 68 m.p.h.				
Train weight	tons	360	360	360
Locomotive weight	,,	177	153	120
Total weight	,,	537	513	480
Diesel engine output	b.h.p.	2,600	2,100	1,700
Maximum speed, 80 m.p.h.				
Train weight	tons	360	360	360
Locomotive weight	,,	240	200	150
Total weight	,,	600	560	510
Diesel engine output	b.h.p.	3,900	3,000	2,300
Maximum speed, 93 m.p.h.				
Train weight	tons	360	360	360
Locomotive weight	,,	310	250	190
Total weight	,,	670	610	550
Diesel engine output	B.h.p.	5,600	4,100	3,100

These figures show that the streamlined train has a low ratio of power per unit of weight, from which it might conceivably be concluded that, as compared with the heavier units, there would be a considerable loss of time when accelerating. That this loss is very small indeed will be seen from Table II, which gives the period of acceleration for the three types of trains from start to 93 m.p.h., and the time required by each to complete the same distance run by the third train as during its period of acceleration.

TABLE II

Train type	1	2	3
Period of acceleration sec.	375	443	520
Distance run during period of acceleration from start to 93 m.p.h. ft.	37,100	43,900	51,200
Final acceleration at 93 m.p.h. ft. per sec. per sec.	0·06	0·06	0·06
Time for covering 51,200 ft. sec.	478	496	520

Comparisons of Different Trains

The factor which finally determines the practical value of the streamline train is economy, and from this point of view no useful purpose is served in comparing trains of ordinary design with the streamline train of the same speeds of either 68 or 93 m.p.h. For instance, at 68 m.p.h. it must at once be evident that a streamline train would not be profitable, while on the other hand at 93 m.p.h. a train of orthodox construction would hardly be a practical proposition, as will be appreciated by reference to Table I. Considerations of economy will therefore be confined in the first place to comparing trains of types 1 and 3 at 80 m.p.h., and afterwards the relative merits of train type 1 at 68 m.p.h. with train type 3 at 93 m.p.h., the brief analysis demonstrating how much more economical a streamline train is at very high speeds, and on the other hand showing that with approximately the same working expenses it is possible to raise the travelling speed some 35 per cent. by introducing a streamline train.

(A) *Comparison of trains with maximum speed of 80 m.p.h.*

The time-table for both trains is practically the same if, for long distances, the small difference in the time for acceleration be neglected. Expense of train staff over a given distance is also equal, and in both cases the locomotives run the same mileage per annum, from which it follows that amounts required for interest and depreciation per mile are in direct proportion to the initial locomotive costs. As regards rolling stock comprising the trains, the costs may be taken as equal for both sets, this assumption being based on the fact that whereas with streamline coaches extra expense would be incurred by reason of the special form of construction, yet on the other hand savings could be effected by simplifying the design of entrance vestibules and windows, the latter being so constructed that passengers could not open them while the train was in motion, thus considerably reducing the frictional air resistance along the sides of the train.

It will be seen from Table I that, as far as locomotives are concerned, at 80 m.p.h. the outputs required are 2,300 b.h.p. for the streamline train and 3,900 b.h.p. for a train of type 1, the capital cost being approximately in the ratio of 1·0 to 1·5, while interest and depreciation on the locomotives would amount to about 8 per cent., which figure has been taken in compiling Table III. These figures, which are based on an annual mileage of 90,000, do not include all working costs, but only compare those items which vary according to the type of train. Even so, the results appear to indicate that there is a saving of about £2,400 p.a. in favour of the streamline train compared with the type 1 unit.

TABLE III.—LOCOMOTIVE WORKING COSTS IN PENCE PER MILE

Train type	1	3
Interest and depreciation	7·7	4·9
Fuel	6·2	3·1
Lubricating oil	1·1	0·6
Total	15·00	8·6

(B) *Comparison of a train of ordinary construction at 68 m.p.h. with a streamline train at 93 m.p.h.*

Locomotive power to haul the ordinary train at 68 m.p.h. will, from Table I, call for 2,600 b.h.p., and at 93 m.p.h. a streamline train would require 3,100 b.h.p., a comparatively small increase in view of the fact that the speed increment is over 36 per cent.

Considered from the point of view of capital cost, the bigger engine would involve an extra expenditure of about 10 per cent., although the charge for interest and depreciation, again taken at 8 per cent., is less per mile for the streamline locomotive. Likewise, as the staff of the high-speed train is, on a mileage basis, better utilised, wage charges—assumed at the rate of 17s. 6d. per day—are accordingly lower. The figures in Table IV are based on a daily run of 280 miles for the ordinary train and 320 miles for the high-speed train, the price of fuel oil being taken, as in the previous table, at 70s. per ton, which is a conservative figure.

TABLE IV.—LOCOMOTIVE WORKING COSTS IN PENCE PER MILE

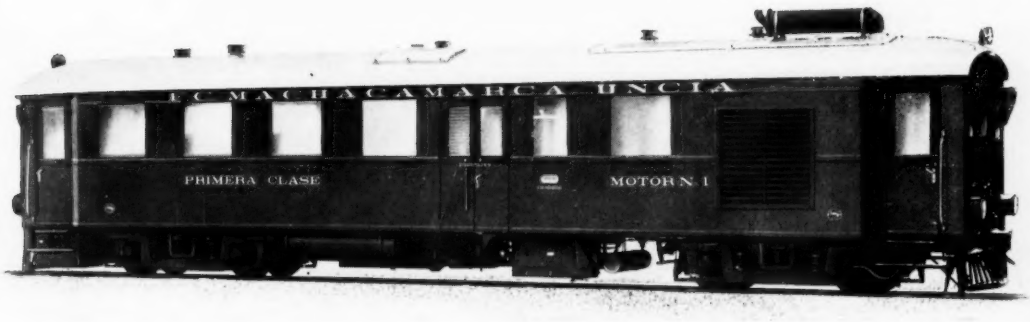
Train type		1	3
Speed	m.p.h.	68	93
Wages		0·8	0·6
Interest and depreciation		6·3	5·4
Fuel		4·6	3·7
Lubricating oil		0·9	0·6
Total		12·6	10·3

Conclusions

It would therefore appear that in spite of the considerable increase in speed, service with the streamline train is cheaper, as also would be the cost of maintenance of the diesel engine per mile. It is probable, however, that maintenance of the running gear for both locomotive and coaches would increase with the higher speed, so that bearing this fact in mind, it appears reasonable to assume that the total costs for the two trains compared will be about the same. The use of the diesel locomotive in conjunction with streamlining may thus open up the way to considerably faster running without increasing working charges, and at the same time the capital cost of the prime mover, due to reduction in power as a result of streamlining, will be definitely lower than for a locomotive with a train of conventional construction, and it is probable that a greater annual mileage will be possible with the high speed train.

DIESELS ON THE ROOF OF THE WORLD

Metre-gauge railcars maintain passenger traffic in the Andes



380 b.h.p. railcar at work in Bolivia

ALTHOUGH the diesel engine, in common with other internal-combustion prime movers, suffers a reduction in power at high altitudes, the operating advantages of diesel railway vehicles are frequently considered sufficient to balance the cost of installing a bigger engine. The most remarkable example of this is to be found in the vehicles described in this article, which maintain a passenger service in the Andes at a height varying from 12,000 to 14,500 ft. above the Pacific.

From the Antofagasta (Chili) & Bolivia Railway's main line the Machacamarca Railway branches off at Oruro, and, after running along the level to Machacamarca, 15 miles, climbs up at a maximum grade of 1 in 40 to 14,500 ft., and then falls to 12,800 ft. at Uncia, the terminal point, 60 miles from Machacamarca. This railway belongs to the Patino Mines Inc., and in addition to the end-to-end passenger service there is a good deal of mineral traffic from the mines in the Huanuni district.

Some time ago two double-bogie diesel-electric railcars were purchased from Germany, and their appearance is shown in the illustration at the head of this article. Seating accommodation for 28 first-class passengers is provided in the car itself, but a 25-ton trailer is hauled, and this carries the third-class native passengers. The total train weight is 70 tons, and is taken up the maximum grade at 12·5 m.p.h. Each of the two railcars makes a daily mileage of 180, and the schedule for a single journey from end to end is 19 m.p.h., the top speed being 37 m.p.h.

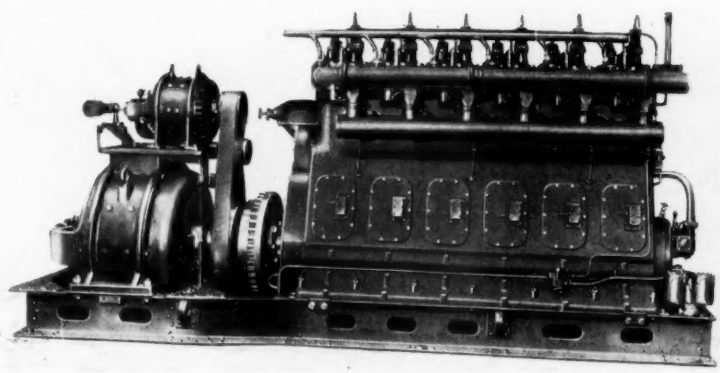
The cars were built by the Triebwagenbau A.G., of Berlin, and are powered by a six-cylinder four-stroke diesel engine built by the Linke-Hofmann-Busch-Werke, which is shown in the second of the accompanying illustrations. The cylinder bore and stroke are 9·5 in. by 13·4 in., and at sea-level the engine is rated at 380 b.h.p. at 650 r.p.m., but including the engine altitude-loss and transmission efficiency only 250 h.p. are available at the rail on the Machacamarca Railway.

Directly coupled to the engine is a 600-volt d.c. generator with an hourly rating of 228 kW., from whose armature shaft the exciter is driven by means of a belt. There are only two traction motors, mounted on the trailing bogie, and they produce an hourly tractive effort at the rail of 4,650 lb. at 12·5 m.p.h. The electrical control embodies a dead-man handle, and a 150 amp. hr. battery is installed for lighting and engine starting purposes.

Braking is on the Westinghouse system, with a supplementary hand brake, and coupling is effected by Henricot automatic centre couplers. The bogies are of the American equalised pattern, with axles running in roller bearings. The principal dimensions of the cars are as follow:—

Gauge	1 metre
Overall length	56 ft. 3 in.
Pitch of bogies	34 ft. 3 in.
Bogie wheelbase	7 ft. 3 in.
Wheel diameter	33·5 in.
Seating accommodation	28
Baggage and mail space	56 sq. ft.
Tare weight	41·5 tons
Maximum speed	37·3 m.p.h.

Special attention has been paid to the cooling of the engine and traction motors in view of the rarefied atmosphere and the 37-mile bank out of Machacamarca.



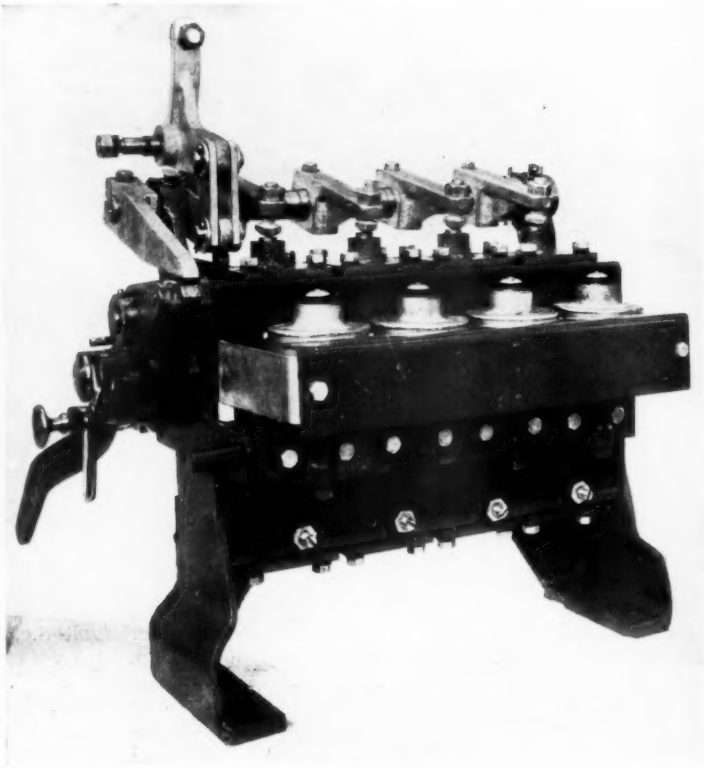
380 b.h.p. Linke-Hoffmann diesel engine

ENGINE GOVERNING MECHANISM

An electro-pneumatic device for diesel-electric locomotives

AN interesting feature of the electrical equipment installed on the three diesel-electric locomotives supplied by the British Thomson-Houston Co. Ltd., early this year to the Ford Motor Company, was the B.T.H. patented governor mechanism mounted on the top of each main generator. This mechanism has been designed to provide a flexible means of remotely-controlling the speed of any type of engine which is governed by the movement of a lever or by varying the tension of a spring. By its use, simultaneous control of two or more engines with several definite engine speeds is readily obtained.

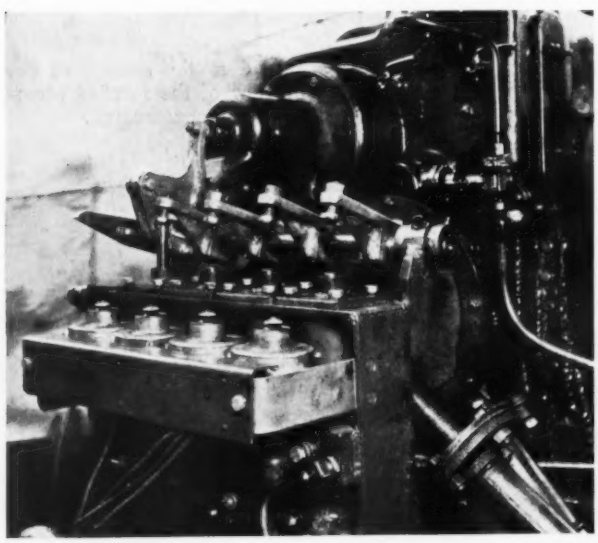
The mechanism shown in the accompanying illustrations is used to obtain five definite engine speeds by varying the engine governor spring tension, thereby controlling the fuel supply to the engine. It consists of a combination of four cylinders, each having a piston, piston rod, and air valve. The piston rods extend through the top covers of the cylinders and each rod engages a separate lever which is attached to a shaft. At one end of the shaft is the throttle lever which engages with a plunger bearing on the governor spring.



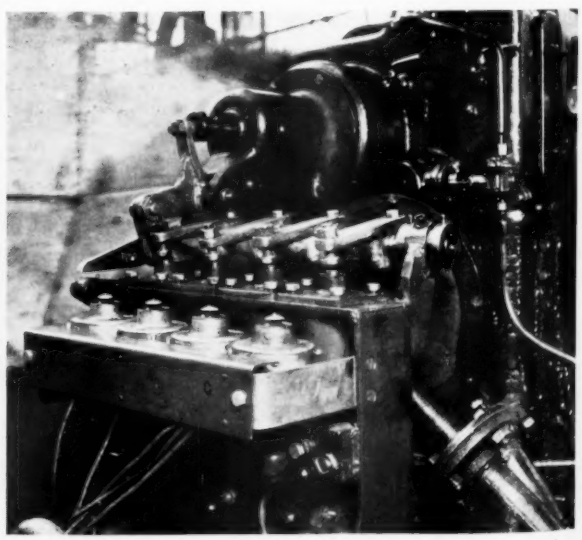
B.T.H. patented electro-pneumatic engine governing mechanism

Each piston rod moves a definite distance; thus the active length of the governor spring corresponds to the movement of the piston rods, and in this way five definite settings of the governor spring, and consequently five definite engine speeds, are obtained. The travel of each piston is limited by a collar fitted beneath the cylinder head, and the pistons and rods are interchangeable.

Adjustment of the effective length of each piston rod is obtained by means of a bolt screwed into the lever which engages with the rod, and the engine speed may be adjusted to suit engine characteristics. Each piston is air-operated, the admission of air to the cylinder being governed by an electro-magnetic solenoid-operated air valve. A strong spring is fitted into each cylinder head in order to return the piston when the air in the cylinder is exhausted, and means are provided for varying the area of the air-valve inlet and exhaust ports in order to control the speed at which the piston operates. The solenoids



Engine running at full speed, all plungers lifted



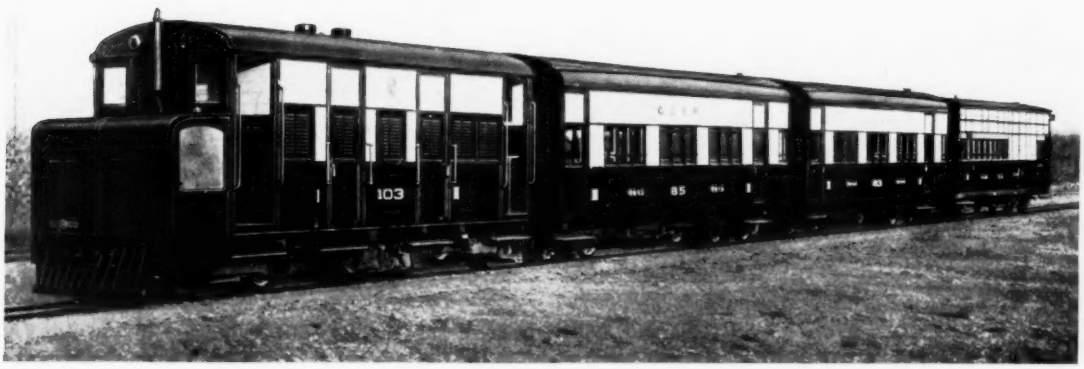
Engine running at idling speed, all plungers released

are energised through contacts on the locomotive master controller, and only one control wire is required for each solenoid where an earth return is used.

The power required to operate the air valves is approximately 10 watts per solenoid. As only two solenoids are energised at the same time, the electrical energy expended is negligible and can be supplied by a small battery when no other source of supply is available. A hand lever is provided in order that the engine may be brought up to full speed when air and electrical power are not available.

The reverse key of the master controller is utilised to provide an effective interlock between the hand lever and the master controller. It is necessary to unlock the hand lever by means of the reverse key before moving the lever from the idling position; the reverse key is in turn locked to the mechanism whenever the hand lever is moved from idling, and the simultaneous control of the engine speed by hand lever and master controller is thus prevented. The interlocking arrangement described above permits the introduction of special features without interfering with any of the approved safety devices used in connection with mechanical transport.

INDIAN RAILCAR RESULTS

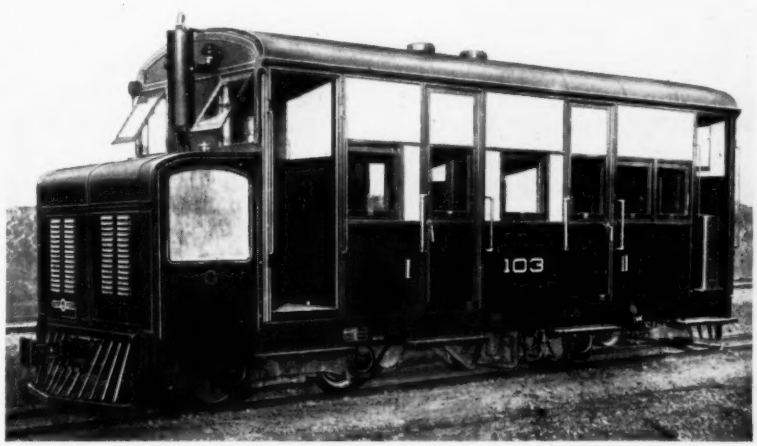


Diesel railcar and train, Baroda State Railways

THE first Armstrong-Whitworth diesel-electric vehicles built for export were the four 95 b.h.p. cars built for the Gaekwar of Baroda's State Railway, which were shipped to India in September, 1932, after trials on the 2 ft. 6 in.-gauge Leek & Manifold Railway. The Baroda State Railway desired to build the bodies in their own shops at Goya Gate, and the accompanying illustrations show the completed car and its normal train.

A full description of the Armstrong-Whitworth portion of these vehicles, including the Armstrong-Saurer engine and A.B.E. transmission, was published in THE RAILWAY GAZETTE for October 7, 1932, so that a recapitulation of the constructive details is unnecessary. All four cars were in service by February 15 last, hauling trains made up of three light trailers, with a combined weight of 30 tons, which are hauled up 1 in 100 grades at 20 m.p.h. The railcar itself is a composite coach carrying two first-class and four second-class passengers, and the three trailers have a capacity for 100 third-class passengers and three tons of luggage.

Up to July 31 last the total mileage of the four cars amounted to 58,573 miles, the average daily mileage per railcar being 94 over the following sections: Goya Gate—Timba Road; Goya Gate—Chota Udaipur; Moti Koral—Chandod; Goya Gate—Padra. We understand that the aggregate mileage is now approximately 80,000, which must be considered very satisfactory in view of the speed limit of 30 m.p.h., and the local train service worked. When the cars first went into service some trouble was occasioned by the cooling arrangements proving insufficient for the hot Indian climate, but modifications were made at the Goya Gate shops with successful results. With a train weight, including the railcar itself, of 43 tons, the fuel consumption has averaged 6·7 m.p.g., or 0·33 gal. per 100 ton-miles on local traffic. 1·75 annas per mile covers the cost of fuel, engine and general lubricating oil, and the driver's wages, while the working expense including interest is 2 annas 5¼ pies per train mile.



95 b.h.p. diesel-electric railcar

LARGE LOCOMOTIVE FOR THE ERIE RAILROAD

Heavy shunting performed at low overall cost by diesel-electric system

THE 800 b.h.p. diesel-electric shunting locomotive delivered to the Erie Railroad by the Ingersoll-Rand and General Electric Companies is one of the heaviest and most powerful diesel shunting locomotives which has yet been constructed, and is actually the largest engaged in railway work. The weight of over 100 tons permits a maximum starting tractive effort of 68,000 lb. to be developed, and while this figure is approached by some of the 600 b.h.p. units built by the two firms mentioned above, the extra horse-power of the Erie locomotive enables the higher range of tractive effort to be maintained to a greater speed.

As may be seen in the accompanying illustration, the locomotive is of the Bo-Bo type of normal American construction. The running gear consists of two equalised trucks, each with a cast-steel frame with the centre cross-member and axle box guides cast integrally. The weight is carried on laminated springs, whose hangers are attached to the equalisers. The underframe is a steel casting, constructed to carry directly the engine-generator set, and has riveted over it a ¼-in. steel floor plate. Each truck is provided with two 12-in. diameter Westinghouse brake cylinders, which operate blocks on the inside of all wheels, and give a total braking force of 75 per cent. of the locomotive weight. The wheels are of the solid rolled type to A.R.A. specifications, and are fitted with bronze hub liners.

The Ingersoll-Rand diesel engine develops its rated output of 800 b.h.p. at 500 r.p.m., and like the 300 b.h.p. model of the same makers it has been developed with a view to complete reliability. The six cylinders have a bore of 14·75 in. and a stroke of 16 in., the maximum fuel consumption at the rated load and speed being 0·45 lb. per b.h.p. hour. The circulating water is cooled in two gilled-tube radiators mounted one on each end of the cab roof.

A G.E.C. 525 kW. generator is directly coupled to the engine, and carries a 55 kW. 125-volt auxiliary generator on a prolongation of the main shaft. The four nose-suspended traction motors, which drive the wheels through single-spur gearing with a ratio of 4·25 to 1, are force-ventilated, one motor-driven blower supplying the air for the two traction motors on each bogie. When passing 1,100 cu. ft. of free air per motor per minute, the tractive effort at the wheel rims is 38,400 lb. at 5·5 m.p.h. on the one-hour rating and 26,000 lb. at 8·3 m.p.h. on the continuous rating. The control system is of the Lemp type, electro-pneumatically operated. Two motors are permanently connected in series, and at starting the four motors are all in series, the second step being provided by a series-parallel grouping, and a third step by shunting a portion of the motor fields.

The principal dimensions and particulars of this Erie locomotive are as follow:—

Wheel diam.	40 in.
Bogie wheelbase	8 ft. 0 in.
Total wheelbase	33 ft. 6 in.
Length between coupler knuckles	46 ft. 4 in.
Maximum speed	40 m.p.h.
Weight in working order	102·5 tons
Maximum tractive effort	68,000 lb.
Fuel capacity	500 U.S. gals.

The following figures relate to a full year's work in Akron yard on the Erie system, as given in the report of the sub-committee on Oil-Electric Locomotives of the American Railway Association.

Number of men in crew	2
Hours in service	6,897
Availability	79 per cent.
Fuel, gals. per hour	6·5
Lubricating oil, gals. per hour	0·1
Working cost per hour, pence (at par)—	
Fuel	13·18
Engine lubricating oil	3·24
Other ,, ,,	0·05
Repairs and maintenance	28·20
Engine-house charges	1·87
Sundries	0·24
Wages	76·00
Net operating cost	122·78
Interest and depreciation	81·90
Gross operating cost	204·68